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Influence of Gyroscopic Moments on the Unbalance Force in Rotor Systems

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***Abstract*- This paper the unbalance excitation force is analyzed to be used as excitation medium in the dynamic characterization of rotors. Analytical and numerical models were used to obtain frequency response Bode and Campbell diagram using unbalance excitation force.**

With the obtained results of the models is concluded that the gyroscopic effects affect the effectiveness of the unbalance force in the excitation of rotors and spite of the force of unbalance does not excite the backward precessional whirl mode in rotors with isotropicbearings; Despite these circumstances, unbalance forceisuseful and easy to apply for the modal characterization of rotordynamic systems.

Keywords—gyroscopic moment, unbalance force, dynamic characterization, unbalance rotors

I. INTRODUCTION

The identification of spatial and modal properties ofa dynamic system is a common practice in modal analysis and rotordynamic [1].

To achieve the identification of dynamic properties, it is necessary toexcite the dynamic systems by a natural input force or a controlled artificial force. In addition, the measure the input and output response signal to be analyzed on the time or frequency domain [2].

When a rotating structureis in operation, the structuralexcitationis the principal problem to apply identification techniques in rotordynamic.

Different excitation devices have been developed and used for real machines o experimental prototypes, these can be preload shaft, magnetic bearing, impact hammer, unbalance of a second shaft, electromagnetic shaker (by an shaft rider) and unbalance force device [3].

The unbalance forces occur on rotating structures for their own operation and can be used for dynamic characterization. The unbalance is the main principal cause of vibration on rotating machines but this onecan also be used to excite a rotating system without connecting it to excitation equipment as electromagnetic exciters or inertial motors.

excitation by unbalance is atechnique very easy to apply and depends onthe rotation speed. The unique disadvantage is that only can be applied in balancing planes [4, 5, 6]. However, the ability of unbalance forceto excitation a rotor depends onfactors such as: the forceapplication point, the gyroscopic effects and bearing characteristics of the rotor.

This work present an analysis of gyroscopic effects and unbalance force in the excitation for rotors in order to dynamic characterization. For thisreason, analytical models and models of finite element of rotors are used to introduce response diagrams and Campbell diagrams using unbalance excitation force.

II. SYNCHRONOUS UNBALANCE EXCITATION

The unbalance force occurs when an eccentric mass spin around a principal axis of inertia and this is the reason of the conversion of rotating energy in passive lateral vibrations. This is a condition of unequal distribution of mass in the radial direction to each section of the shaft. This establishesa condition of unbalance in the center of mass of the rotor, which does not coincide with its axis of rotation.

If the first lateral mode of vibration in an isotropic rotor is considered then the unbalance distribution may be considered as an average of a force composed of a mass ***m*** at a radius ***r***. The magnitude of the unbalance force depends directly on the speed of rotation of the body and it is expressed as:

$$F_o = mr\Omega^2 \tag{1}$$

The magnitude of unbalance force can be represented as the addition of components of one vector around each reference axisand it can be expressed as:

$$|F| = \begin{Bmatrix} F \\ 0 \end{Bmatrix} \cos \Omega t + \begin{Bmatrix} 0 \\ F \end{Bmatrix} sen\ \Omega t \tag{2}$$

$$\begin{aligned} F_z &= F cos\ \Omega t \\ F_y &= F sen\ \Omega t \end{aligned} \tag{3}$$

WhereF_y and $F_z$are components of the unbalance force to each axis.

The unbalance excitation force can also be represented using a harmonic excitation force within movement equation as:

$$M\ddot{z} + C\dot{z} + Kz = mr\Omega^2 e^{j(\Omega t)} \qquad (4)$$

where M, C and K represent the mass, damping, and stiffness matrices respectivelywhich the right hand side of the equation (4) represents the generated unbalance force by the eccentricity of one rotatory element. The unbalance force is a synchronous excitation where the excitation frequency is equal to the frequency of rotor speed.

The characteristics of the bearing stiffness are important factors when using excitation by unbalance. If the bearing stiffness of a rotor has the same value in each principal directionsthen it is called 'isotropic rotor' with respect to the bearings. If the bearing stiffness of a rotor has the different values in the principal directions then it is called 'anisotropic rotor' with respect to the bearings.

The isotropy or anisotropy with respect to the bearing stiffness in a rotor affects directly the response of excited system. For example, for the case of an isotropic rotor, the unbalance force will only excite the vibration modes of positive precession or forward whirl (Fw) and will not excite the vibration modes of negative precession or backward whirl (Bw). This is because there is no energy exchange between negative precession mode $\{\phi_1\}$ and unbalance force$\{F\}$; this means that the vector of unbalance force and vector of negative precession mode are orthogonal. The otherwise is when the negative precession mode $\{\phi_2\}$ is not orthogonal to the unbalance force$\{F\}$.

In an anisotropy rotor, the unbalance force will excite two vibration modes (the negative precession mode as the positive precession mode). To illustrate this, we will present the following numerical model (Fig.1). In this figureis possible to see the placing of unbalance mass and the node where the response is measured. For this case, a rotor cantilever with isotropic and anisotropic bearingsis modeled.

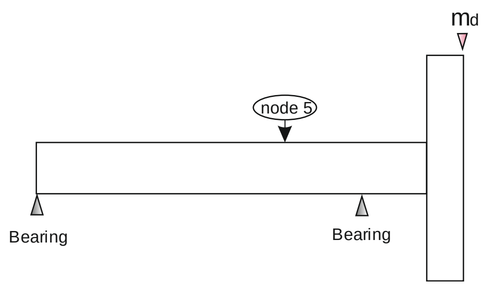

Fig. 1. Cantilever rotor model

A harmonic analysis was realized for this model in order to obtain the Campbell diagram and unbalance frequency response using the properties of an anisotropic and isotropic rotor.

The Campbell diagram is showed in the Fig. 2 where each crossing with $1x$ represents a critical speed of the rotor, these crossings must be identifiedin the response diagram like resonance peaks.

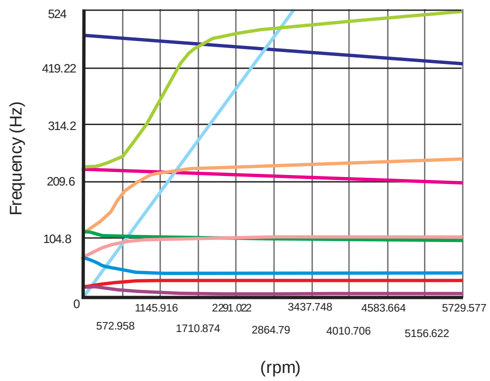

Fig. 2. Campbell diagram

The response diagrams of rotor to isotropic and anisotropic case are shown in the Fig. 3. The blue line (-o-) is the response of rotor with anisotropic case and red line (---) is the response of rotor with isotropic case.

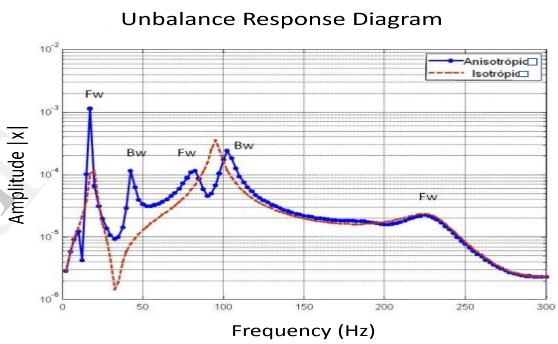

Fig. 3. Unbalance response: isotropic and anysotropic case

The response of anisotropic system shows resonance peaks for the modes of positive (Fw) and negative (Bw) precession. In the opposite case (isotropic system), the response diagram only shows the resonance peaks for modes in positive precession.

The fact of that the unbalance force does not excite the negative precession modes for the characterization case of a rotor-dynamics system is a disadvantage because it is not possible to identify the critical speeds and damping corresponding to these vibration modes. Although, unbalance force does not excite the modes of negative precession, these modes are intrinsic properties of the rotor and may be excited in operation by other forces producing faults in resonance or instability in the rotor.

III. EFFECT OF GYROSCOPIC MOMENT ON THE UNBALANCE EXCITATION

Most rotary systems are under the influence of forces known as gyroscopic forces. These forces occur predominantly at high speed; their magnitude dependson the rotational speed and inertial properties of the system, which sometimes become negligible [7].

The forces produced by gyroscopic moments, which are created when a rotor o inertial element is rotating around of its principal axis of inertia and tends to rotate in any of its other principal directions.

Gyroscopic moments are manifested in the elements that have concentrated mass or disk shape in a rotating system and the magnitude of these is represented analytically by the following equation:

$$\begin{Bmatrix} m_x \\ m_y \\ m_z \end{Bmatrix} = \Omega \begin{bmatrix} 0 & I_p & 0 \\ I_p & 0 & 0 \\ 0 & 0 & 0 \end{bmatrix} \begin{Bmatrix} \dot{\theta}_x \\ \dot{\theta}_y \\ \dot{\theta}_z \end{Bmatrix} \qquad (5)$$

where:

I_p Polar moment of inertia of a disk

Ω Rotational speed

$\dot{\theta}$ Velocity of precession

In the equation of motion of a rotating mechanical system, the gyroscopic effects are within the damping matrix producing that the matrix became an asymmetric matrix and therefore coupling the degrees of freedom of the system [8]. In addition, another phenomenon that occurs with the gyroscopic moments is the increased of forward-mode stiffness and decreased backward-mode stiffness in the rotor. With respect to the foregoing, the natural frequencies and the critical speeds of the system change with respect to the rotational speed.

The distribution of the masses or rotating elements along the length of the rotor or shaft is important in order to influence of the gyroscopic effects, since the magnitude of the gyroscopic moment in a rotating element or disk depends on the degree of tilt or rotation relative with respect to principal axis of inertia.

Gyroscopic moments are predominant in the case of flexible modes and only for conical modes in rigid rotors (angular displacements). These effects cause angular displacements on inertial disks where unbalance masses are applied producing that the unbalance force to decompose into smaller amplitude forces affecting the effectiveness of the unbalance force in the excitation of a rotor. In the previous section, the components of the forces of unbalance in the plane perpendicular to the mode shape were established in the Equation (3), which depend on the angle of force.

An important quality of the exciting force from a physical perspective is that the perpendicular forces to modal shapes can be better excited in a rotating structure; from a spatial perspective an orthogonal force vector to the modal shape will not cause excitation, nevertheless a collinear force vector to the mode shape excite it better.

In accordance with the above, it is possible toconclude that to obtain a better excitation from the point of view of the force magnitude, the force must be placed in a plane perpendicular to the modal shape.

Now, we will consider the case where a known unbalance force, which is outside of perpendicular plane. The Fig.4 represents the unbalance force in the space.

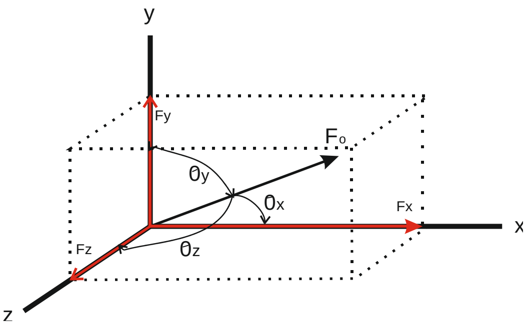

Fig. 4. Unbalance force decomposition.

The components of this force to the **zy** plane correspond to plane perpendicular to the mode shape and are expressed as:

$$\begin{aligned} F_z &= F\cos\Omega t\,.\cos\theta_y \\ F_y &= F\sin\Omega t\,.\cos\theta_z \end{aligned} \qquad (6)$$

The previous equations show the rectangular components of the unbalance force, now the components do not depend only on angular displacement in the plane ***zy***but also in others angular displacements.

Therefore, the magnitude of the unbalance force in the plane perpendicular to the modal shape will be much smaller than without angular displacement on ***xy*** and ***zx***planes.

$$\begin{vmatrix} F_z = F_o\cos\Omega t \\ F_y = F_o\sin\Omega t \end{vmatrix} > \begin{vmatrix} F_z = F_o\cos\Omega t\cos\theta_y \\ F_y = F_o\sin\Omega t\cos\theta_z \end{vmatrix} \qquad (7)$$

For example, we consider the equation of motion (4) to rotor model presented in [1] with the following spatial properties.

$$\begin{bmatrix} 8.03 & 0 \\ 0 & 8.03 \end{bmatrix}\ddot{z} + \begin{bmatrix} 0.1 & 1.09 \\ -1.09 & 0.1 \end{bmatrix}\dot{z} + \begin{bmatrix} 994 & 0 \\ 0 & 600 \end{bmatrix}z = \begin{bmatrix} -0.028 & 0 \\ 0 & -0.028 \end{bmatrix}e^{j(\Omega t)}$$

Two cases were considered for the analytic solution of previous equation. In the first instance, the unbalance force is maintained in the radial direction and in the plane perpendicular to the axis of rotation of the rotor. As follow:

$$\textbf{\textit{First}}F_1 = \begin{bmatrix} F\cos\Omega t = 0.028 \\ F\sin\Omega t = 0.028 \end{bmatrix};$$

In the second instance, the unbalance force is assumed with angular displacement around to the others axes using $\theta_y\ and\ \theta_z = 30°$:

$$\textbf{\textit{Second}}F_2 = \begin{bmatrix} F\cos\Omega t.\cos 30 = 0.024 \\ F\sin\Omega t\,.\cos 30 = 0.024 \end{bmatrix}$$

If we consider a harmonic solution to the equation of motion, the response of dynamic system is shown in Fig.5.Where the red curve represents the first case and the blue curve plot the second case. This Figure shows the existence of a variation in the amplitude of the response for each case, it is mainly due to tilt or angular displacement of the force in other planes as in the second case.

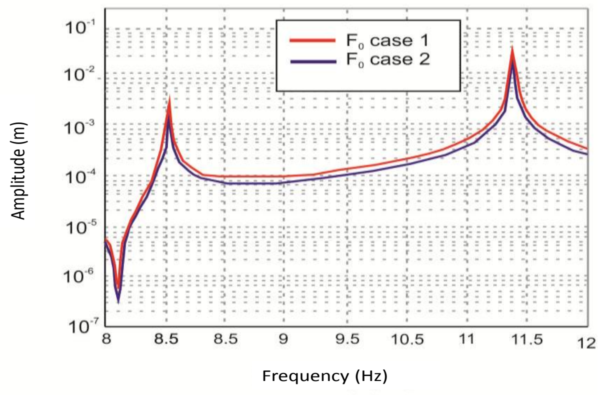

Fig. 5. Synchronous unbalance response shifts

Thus, the magnitude of the response to unbalanced depends mainly on how well the vibration modes are excited.

In the Fig. 5, the magnitude of the unbalance force increases or decreases depending directly with the rotational speed and so a better excitation is expected in higher modes due to the magnitude of the force. Among higher the rotation speed is more significant is the gyroscopic effect which causes the discs, concentrated masses and rotor elements have angular displacements, which causes unbalance force decompose into smaller magnitude forces as explained above.

IV. NUMERICAL SIMULATION

To analyze the rotor unbalance excitation with more components and complexity were developed a finite element model.

Rotordynamic-systems analysis involves the study of some variables related to the vibration as the natural frequencies, critical speeds, mode shapes or full system response to load imbalance and instability of the components during vibration.

The main goal of this section is to calculate the critical speeds, modal shapes, response by unbalance, and from previous we can evaluate the effect of gyroscopic moment on excitation by unbalance.

Fig.6 shows the rotor model used.In this case, a rotor is modeled, which has three discs and changes in diameter. The rotor is anisotropic with respect to the supports and the mass imbalance is located on a disc as shown in Figure 8. The operating speed of the rotor is $380\,\frac{rad}{s}$.

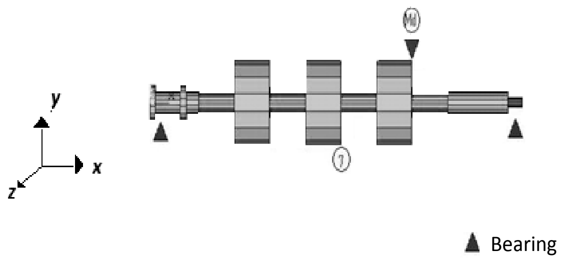

Fig. 6. Rotor system modeled

The rotor characteristics and bearing properties are showed in Table I

TABLE I.Natural frequency, critical speeds and rotor properties

Parameters	Natural Frequency (Hz)	Critical Speeds (RPM)
$Kzz = 4x10^8\,\frac{N}{m}$	22.13 22.15	Bw 1320 Fw 1336
$Kyy = 8x10^8\,\frac{N}{m}$	98.17 98.45	Bw 5526 Fw 6353
$Czz = 5000\,\frac{Ns}{m}$	243.14 244.14	
$Cy = 10000\,\frac{Ns}{m}$	*Fw*= forward-mode	
$L = 1.33m$	*Bw*= backward-mode	

A modal analysis and unbalanced harmonic analysis are undertaken for this model. The results are shown in the Fig.7. This oneshows a diagram of the rotor unbalance response, where we identifythe resonance peaks corresponding to four excited vibration modes. As the rotor is anisotropic, the unbalance force excites the twotypes of modes (*Fw* and *Bw*).

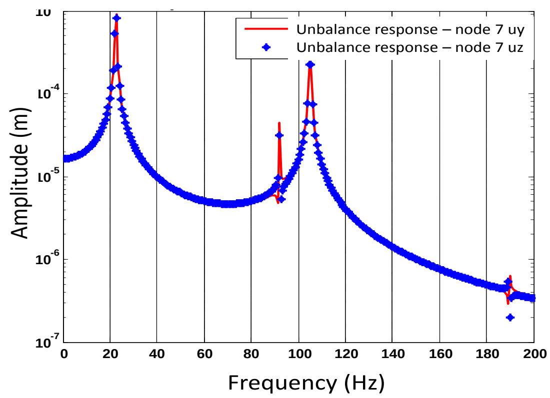

Fig. 7. Unbalance response -node 7

The results of Campbell diagramare given in the Fig. 8, which shows the variation in frequencies and Fig. 9 shows the logarithmic decrement coefficient (damping) in the system regarding the speed of the rotor caused by the gyroscopic moments.

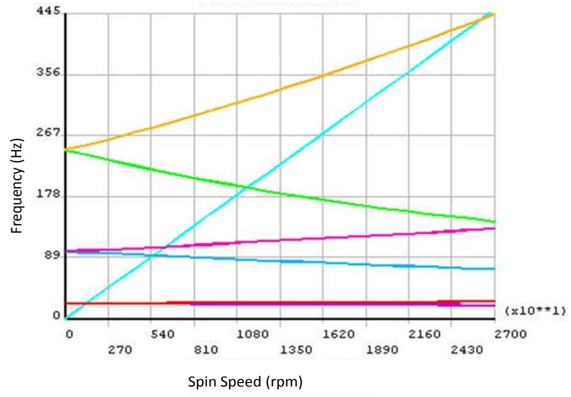

Fig. 8. Campbell Diagram

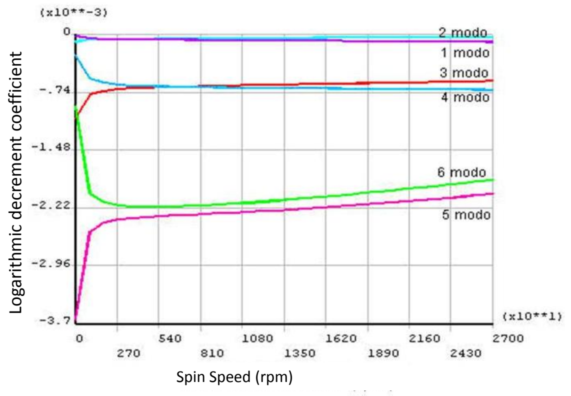

Fig. 9. Campbell Diagram(damping)

Themode shapes for the rotor are shown in Fig. 10. As seen in the mode shapes, the discs have a greater angular displacement giving results of gyroscopic moments with a higher magnitude.

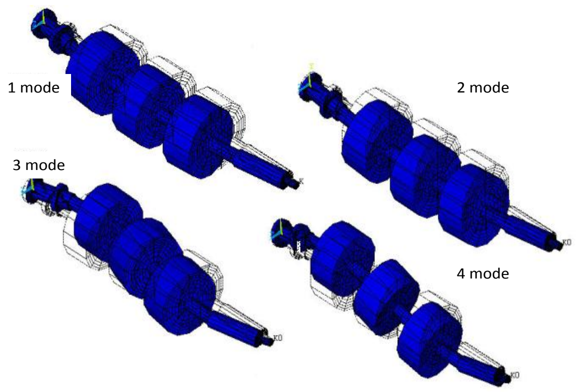

Fig. 10. First four mode shapes

The response using unbalance excitation is affected by the following parameters: the excitation point, mode shapes, damping and gyroscopic effects inherent in the rotor for numerical case.

The first two modes of vibration have 22 Hz and 22.26 Hz and are semirigid modes, which arecharacterized mainly by translational movements. These modes are very little affected by gyroscopic moments as seen in the Campbell diagram at the first intersection, when rotational speed is increased, the natural frequencies change very few.

In the logarithmic decrement diagram(see Fig. 9) shows that the energy dissipation or damping for the first vibration mode increases and decreases for the second vibration mode. This phenomenon is reflected in the response pattern, the first two modes have resonances with greater amplitude. The contribution of the second vibration mode is small due to the close modes.

The third and fourth mode of vibration are at 92.1 Hz and 105.8 Hz,these ones are flexible modes characterized by angular movements andthe gyroscopic moments are predominant in the second crossing andnatural frequencies vary significantly increasing the rotational speed causing the separation of frequencies(see Campbell diagram).

The logarithmic decrement diagram shows that the damping for the third mode decreases and increases for the fourth mode vibration.

V. CONCLUSIONS

The unbalance force is an alternative for excitation of rotors in order toidentify or characterize oftheir dynamic,since these forces are present in all rotary system and their magnitude can be known in order to control the unbalance mass, which is placed at the balancing of rotors.

The unbalance force can only be applied in balancing planes and can excite only modes of positive precessing in isotropic rotors. This could be a drawback at using unbalance excitation. However, taking into account the lack of homogeneity of materials, finishes and manufacturing processes, often there will be some degree of anisotropy that excite the negative precession modes.

Unbalance excitation is affected by gyroscopic moments, the principal reason for thisis because the unbalance is mainly placed on the disks or concentrated masses causing the decomposition of the magnitude of the unbalance force.

In order to evaluate the response of the system, the numerical model discriminates or distinguish between the effects of damping, excitation point and gyroscopic moment.

Gyroscopic momentsaffect the unbalance excitation. This one isindependent of changes of damping. However, these effects arerelated withthe location of the excitation point in the system.

Thus, this work introduced a new variable that affects unbalance excitation (gyroscopic moments) and should be taken into account in order to plan experimental identification tests or dynamic characterization of rotating structures.

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